

INVESTIGATION OF SYSTEMS AND TECHNIQUES  
FOR MULTICOMPONENT MICROFORCE MEASUREMENTS  
ON WIND TUNNEL MODELS

Semiannual Status Report  
on  
NASA Grant NGR 47-005-026

Submitted by Co-Principal Investigators:

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## FOREWORD

This is a preliminary report of work done on NASA Grant No. NGR-47-005-026 during the six month interval ending December 31, 1966. The information presented is not intended to serve as a final reference for design information; final results are a few months from completion on all phases of the work in progress. The information presented does indicate the present state of the effort and the direction in which the work is moving. All useful technical data will be repeated in a final report.

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## SECTION I

### INTRODUCTION

Most of the effort in the past six months has been directed toward the development of bearings for a rocket engine test stand, considerations for calibration of the test stand and manipulation of data. A detailed theoretical analysis is underway to support these efforts. Subsidiary consideration of test stand design, in general, has been required too. The construction and partial testing of a high temperature motor sensor-flexure assembly for a skin friction balance has been accomplished. The data are included in this report.

The purpose of the rocket engine test stand work is to aid NASA in the development of a test stand which will make possible testing to frequencies in the neighborhood of 100 Hz. All components of thrust are to be measured which result from thrust vector control experiments.

The rocket engines to be tested will weigh approximately 2000 pounds, have a length of approximately 5 feet and have a diameter of 3 feet.

The most severe problems associated with rocket engine test stand work are due to the thermal growth of the engine and the dynamic characteristics of the test stand including force sensors--also called load cells. Air and/or hydraulic bearings are being developed to support one end of the rocket so that thermal growth will not introduce forces in the sensors. Test stand dynamic problems are to be minimized through construction. The structure will be made very rigid and short load cells having very high spring constant are to be used. However, mechanical resonances inside the frequency range of interest, to 100 Hz, are anticipated. Therefore, research on calibration and data manipulation is in progress to minimize errors introduced by the imperfect test stand dynamics.

The major areas of effort are described in detail in the following separate sections.

## SECTION II

### BEARING DESIGN AND TESTING

Modifications to the existing air bearing to adapt it for use as a hydraulic bearing have been determined. However, before physically modifying the bearing, it will be tested as an air bearing, operating with sufficient gas supply pressures to support the same test forces as the hydraulic bearing.

As it was necessary to move the laboratory test area, a new test stand to support the bearing and associated test equipment has been built. It is now ready to have the bearing installed.

#### A. Introduction

Low friction fluid bearings will be used in the rocket test stand to decouple the measured force components. These bearings must support the test stand "dead" load, as well as the rocket thrust forces, and must operate satisfactorily for large variations in loading. In addition, they must have a large spring constant, to keep the test stand resonant frequencies as high as possible, and sufficient damping.

Because of the large support forces (approximately 2000 lbs.) and the requirements of large spring constant and damping coefficient, a hydrostatic bearing using oil as the support fluid was selected. The existing air bearing [1] could be modified for use as an oil bearing. However, before the irreversible modifications were made, it was decided that the air bearing should be tested for possible use in the test stand.

If air bearings could be used, they would have the advantage of simple operation with respect to the support fluid supply: Oil bearings require an oil pump, a sump to recover the oil passing out of each bearing and oil pressure regulators. The supply for an air bearing need only be a bottle of compressed nitrogen with suitable regulators, the nitrogen

escaping to the atmosphere after passing through the bearing. The disadvantages of the air bearing are its closer tolerances and thus, greater susceptibility to damage, and its lower damping than the oil bearing.

Aside from the advantages and disadvantages of the air bearing with respect to the oil bearing, it is important that the air bearing performance evaluation be extended while the bearing is still available. Evaluation of the properties, and improvements in performance, of the air bearing can be directly applied to the wind tunnel test apparatus.

Since the same test facility can be used to test both the air and oil bearings, little additional time will be required to test the air bearing prior to its modification. In fact, the only additional time is that necessary to fabricate the capillary flow restrictors, assemble the bearing and perform the tests.

Both static and dynamic tests will be used to evaluate the performance of the bearings. Static load vs. displacement measurements will be made to determine the static spring constant. The damping will be determined by using an electromagnetic "shaker" motor to apply dynamic forces to the bearing. The basic test stand to hold the bearing and shaker has recently been completed. An apparatus to statically load the bearing is being designed.

#### B. Air Bearing

To prepare the air bearing for testing it was necessary only to fabricate new capillary flow restrictors. An optimum length was determined for the capillaries, to maximize the bearing spring constant. No effort was made in the earlier work to choose an optimum length for the capillaries; the length used was selected on the basis of convenience [2] .

The flow of gas into a single capillary is given by

$$Q_s = \frac{\pi r_c^4}{8\mu l_c} (P_s - P_o) \quad (1)$$

where

$Q_s$  = volume flow of gas through capillary,  $\text{in}^3/\text{sec}$ ;

$r_c$  = radius of capillary in inches;

$l_c$  = length of capillary in inches;

$P_s$  = absolute capillary source pressure,  $\text{lb}/\text{in}^2$ ;

$P_o$  = absolute capillary exhaust pressure,  $\text{lb}/\text{in}^2$ ;

$\mu$  = dynamic viscosity,  $\text{lb-sec}/\text{in}^2$ .

The flow out of one quadrant of the bearing pad, supplied by one capillary is

$$Q_e = \frac{bh^3}{12\mu l} (P_o - P_e) \quad (2)$$

where

$e$  = volume flow of gas out of quadrant,  $\text{in}^3/\text{sec}$ ;

$b$  = perimeter of pad, inches;

$h$  = gap height between pad and supported member, inches;

$l$  = length of land, where pressure drop  $P_o$  to  $P_e$  occurs, inches;

$P_o$  = absolute pad pressure,  $\text{lb}/\text{in}^2$ ;

$P_e$  = absolute exhaust pressure,  $\text{lb}/\text{in}^2$ .

Equations (1) and (2) may be written in terms of gauge pressure, capillary diameter,  $d_c$ , and for  $P_e$  equal to atmospheric pressure:

$$Q_s = \frac{\pi d_c^4}{128\mu l_c} (P_s - P_o) \quad (3)$$

$$Q_e = \frac{bh^3}{12\mu l} P_o \quad (4)$$

Since the flow through an orifice is equal to the flow out from one quadrant, equations (3) and (4) may be set to be equal, giving

$$P_o \left( \frac{k_Q h^3}{12\mu} + \frac{\pi d_c^4}{128\mu l_c} \right) = \frac{\pi d_c^4}{128\mu l_c} P_s \quad (5)$$

where  $k_Q = b/l$ .

If the supported weight per quadrant is  $w$ , then the spring constant,  $s$ , is

$$s = \frac{dw}{dh} \quad (6)$$

or since  $w = AP_o$ , where  $A$  is the effective pad area for one quadrant, then

$$s = A \frac{dP_o}{dh} = AS.$$

Here,  $S = dP_o/dh$  is the spring constant per unit area of the pad.

Taking the appropriate derivative from equation (5) gives

$$S = - \frac{K_c k_Q P_s}{4\mu} h^2 \left( \frac{k_Q h^3}{12\mu} + K_c \right)^{-2} \quad (7)$$

where

$$K_c = \frac{\pi d_c^4}{128\mu l_c} \quad (8)$$

The maximum static spring constant may be found by letting  $ds/dh = 0$ , so from equation (7) the condition

$$K_c = \frac{k_Q h^3}{6\mu} \quad (9)$$

results. Substituting for  $K_c$  from equation (8)

$$\frac{d_c^4}{l_c} = \frac{128k_Q h^3}{6\pi} \quad (10)$$

is the condition for maximum static bearing stiffness.

The physical dimensions of the bearing quadrant are:

$$b = 7.43 \text{ inches}$$

$$h = 5 \times 10^{-4} \text{ inches}$$

$$l = 0.125 \text{ inches}$$

Substitution of these values into equation (10) gives

$$l_c = \frac{d_c^4}{5.05 \times 10^{-8}} \quad (11)$$

from which the following table for hypodermic tubing may be determined.

| GAUGE | $d_c$<br>inches      | $l_c$<br>inches | $20d_c$<br>inches |
|-------|----------------------|-----------------|-------------------|
| 25    | $10^{-2}$            | 0.198           | 0.20              |
| 24    | $1.2 \times 10^{-2}$ | 0.412           | 0.24              |
| 23    | $1.3 \times 10^{-2}$ | 0.566           | 0.26              |
| 22    | $1.6 \times 10^{-2}$ | 1.30            | 0.32              |
| 21    | $2 \times 10^{-2}$   | 3.17            | 0.40              |

To insure laminar flow through the capillary, the condition that  $l_c > 20d_c$  was chosen. It may be seen from the table that all hypodermic gauges with the exception of gauge 25 fulfill this requirement. Gauge 24 was chosen, the others being too long to be mounted in the bearing.

The required pad pressure to support a weight,  $w$ , per quadrant is

$$P_o = \frac{w}{A}; \quad (12)$$

again  $A$  is the effective pad quadrant area. Taking  $A = 3.36 \text{ in}^2$ , the area of constant pressure [3], then  $P_o = 74.5 \text{ psig}$  for a supported weight of 250 lb. per quadrant or 1000 lb. total load.

Equation (5) may be used to compute the supply pressure,  $P_s$ . Solving for  $P_s$  gives

$$P_s = \left( \frac{kh^3}{12} + \frac{d_c^4}{128l_c} \right) \frac{128l_c}{\pi d_c^4} P_o \quad (13)$$

For maximum bearing stiffness and a supported weight of 250 lb. per quadrant ( $P_o = 74.5 \text{ psig}$ ),  $P_s = 112 \text{ psig}$ .

Finally, the flow rate may be computed from equation (4). Upon proper substitution,  $Q = 17.8 \text{ in}^3/\text{sec}$  per quadrant. This gives a total flow rate of  $71.2 \text{ in}^3/\text{sec}$  for the entire bottom bearing pad.

### C. Bearing Modifications

To modify the existing gas bearing for use with oil as a hydrostatic bearing, it is necessary only to deepen the distribution grooves [4] and change the flow restrictors to the proper size. The bearing design was based on the series of two papers by Boyd, Kaufman and Raimondi [5]. Although calculations for both the unopposed and opposed pads were made, only the opposed pad design is described here and is based on part II of the above mentioned paper. Definitions of pertinent variables used in the design equations follow:

$\beta_c$  = capillary design parameter, dimensionless;

$\alpha_c$  = film thickness factor,  $\text{in}^{-1}$ ;

$C$  = half clearance (gap height), in;

$e$  = displacement of opposed pad from mid-position, in;

$\epsilon$  =  $e/C$  = relative displacement, dimensionless.

The defining equation for  $\beta_c$  is

$$\beta_c = (\alpha_c^3 C^3)^{-1} \quad (14)$$

where

$$\alpha_c = \left( \frac{32k l_c}{3\pi d_c^4} \right)^{1/3} \quad (15)$$

For maximum stiffness  $\beta_c \approx 1$ , for a wide range of displacements [6]. Since  $C$  is known, equations (14) and (15) may be used to determine the ratio

$$\frac{d_c^4}{l_c} = \frac{32k_Q C^3}{3\pi} \quad (16)$$

From this equation a table for standard hypodermic tubing may be determined:

| GAUGE | $d_c$<br>inches      | $l_c$<br>inches | $20d_c$<br>inches |
|-------|----------------------|-----------------|-------------------|
| 19    | $2.7 \times 10^{-2}$ | 0.168           | 0.54              |
| 18    | $3.3 \times 10^{-2}$ | 0.375           | 0.66              |
| 17    | $4.2 \times 10^{-2}$ | 0.980           | 0.84              |
| 16    | $4.7 \times 10^{-2}$ | 1.54            | 0.94              |
| 15    | $5.4 \times 10^{-2}$ | 2.68            | 1.08              |
| 14    | $6.3 \times 10^{-2}$ | 4.98            | 1.26              |

Here, the total gap height,  $2C$ , has been taken to be  $5 \times 10^{-3}$  inches. Again, from the requirement that  $l_c > 20d_c$  for laminar flow through the capillary, tubing gauges larger than 17 are not satisfactory. This means that the minimum hypodermic tubing length for maximum stiffness is 0.98 inches. Shorter lengths with corresponding smaller diameters will reduce the stiffness.

If the weight to be supported is 250 lb per quadrant, the pad pressure must be 74.5 psig. For an absolute displacement of  $e = 10^{-3}$  inch, the relative displacement is

$$\epsilon = \frac{e}{c} = 0.4 .$$

This relative displacement will reduce the stiffness for  $\beta_c = 1$  by a factor of 0.766, as determined from figure 2-c in reference [5] . Then from figure 2-a of the same reference

$$\frac{P_2 - P_1}{P_s} = 0.55 \quad (17)$$

for  $\beta_c = 1$  and  $\epsilon = 0.4$ , where  $P_2 - P_1$  is the differential pressure between the upper and lower pads. This differential pressure must equal to that to support the bearing load; in this case  $P_2 - P_1 = 74.5$  psig. From equation (17) the source pressure is  $P_s = 135.5$  psig.

Also, from figure 2-b, reference [3]

$$\frac{12\mu}{k C^3 P_s} Q = 0.9 \quad (18)$$

from which it may be determined that the flow rate,  $Q$ , is  $2.14 \text{ in}^3/\text{sec}$  per quadrant or  $0.558 \text{ gal/min}$  for an oil with a viscosity  $\mu = 4.4 \times 10^{-6} \text{ lb-sec/in}^2$ .

The power requirements are determined from

$$H = \frac{P_s}{6600} \quad (19)$$

where  $H$  is the pumping power in horsepower. With the above determined flow rate and pad pressure the total power for four quadrants is  $0.176 \text{ hp}$ .

In addition to deepening the distribution grooves, and installing proper capillaries in the bearing for use with oil, a sump, to receive the oil flowing out of the bearing must be provided. Proper pressure will be provided by a small pump and pressure regulators.

D. Bearing Tests

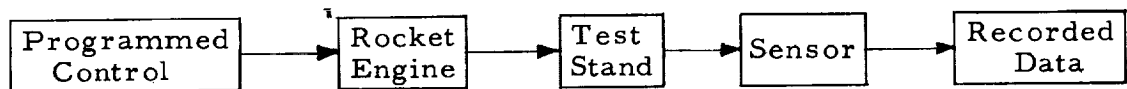
Both the air bearing and the oil bearing will be tested in the same manner, in the same test stand. A new concrete test stand has been built to accommodate the bearing and test apparatus. The supported or floated member of the bearing is connected to the armature of an electromagnetic type "shaker" motor, to be used for dynamic tests. This connection also prevents the supported member from moving laterally across the pad.

Static loading of the bearing will be done by weights hung from the bearing by a balance arm arrangement. Initially, only downward loading will be used, as it is the properties of the bottom pad which are of interest. This is because a top or hold-down pad of proper design is not available; the hold-down pads do not have distribution grooves, to create an area of constant pressure. There is the disadvantage that the effective mass of the system is increased for the dynamic tests.

### SECTION III

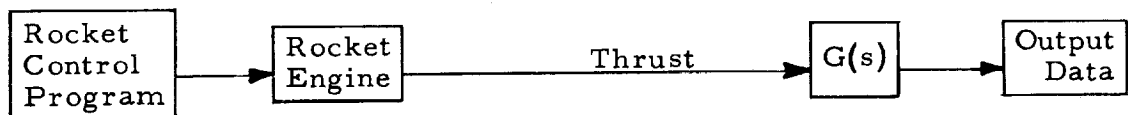
#### TESTING AND DATA MANIPULATION

Test stand data for rocket engines is usually obtained by mounting the rocket on a test bed and measuring components of thrust by use of strain gauges mounted inside load cells. This system can be represented by the model



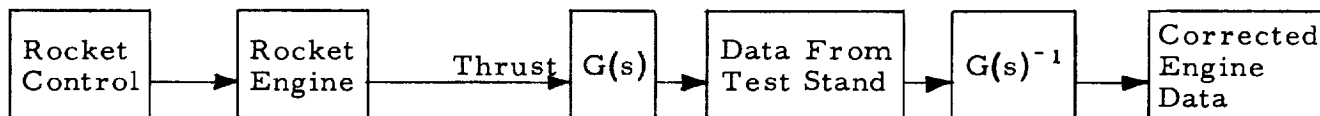
Ideally, the test stand-sensor combinations should not modify the data, i. e., the data are ideally directly proportional to the thrust of the rocket engine. This requires that the test stand not store energy in the range of frequencies used for the test. However, the test stand-sensor combination does affect the signal because mechanical resonances occur in the frequency range of interest. The removal of this effect using analog methods is the purpose of this part of the investigation. Two possible solutions to the problem are: 1) determine the test stand transfer function from which the data may be modified to separate engine performance from test stand response; 2) exclude from the test program and possibly the data those frequency components in the range of test stand resonances.

Suppose the test stand-sensor combination, including the engine weight and mechanical characteristics, has a transfer function  $G(s)$ . The system will be as shown:



This model has two unknowns, the rocket response (which is the desired quantity to be measured) and the transfer function  $G(s)$  whose effect is to

be eliminated. Actual engine data can be obtained by numerical calculations or by an analog approach which would allow on-line correction of data, shown schematically as:



In this approach the corrected data are directly proportional to the thrust because the test stand data have been put through a network whose characteristic is the inverse of the test stand. The network  $G(s)^{-1}$  must be physically realizable for this system over a sufficiently large bandwidth to make this data manipulation scheme practical. Fortunately, this should not be a problem. The major difficulty involved is to find  $G(s)$ , since not only is  $G(s)$  time dependent, but data about the nature of  $G(s)$  are not very repeatable. However, it must be assumed that  $G(s)$  is known for any kind of data manipulation including the numerical approach.

It has been shown by Sprouse and McGregor [7] that analog methods can be used for on-line data reduction. The model used for work done by Sprouse and McGregor was time invariant and of second order. From their model the basic method can be extended in two ways: 1) to more complex systems of higher order; 2) time variant systems.

Extension to higher order systems takes into account cross coupling between the applied forces of course, requires experimental data to obtain the model constants. The data obtained from other workers in this area all indicate that even data taken under good conditions is of little use in accurately determining the transfer function. Data recorded during test for system identification would be expected to be even worse than this unsatisfactory data. Hopefully, one may not need an accurate representation for the second order effects associated with cross coupling because this should be a small part of any total component.

Emphasis is presently being placed on resolving the question of how accurately the test stand must be described to obtain final engine data of a given accuracy. This particular work is being supported by investigation of a theoretical model of a six degree-of-freedom system to predict the cross coupling for  $G(s)$ .

The mass and center of gravity change as the fuel is consumed. The resultant effect on  $G(s)$  must be taken into account. Some of the methods under consideration use variable resistors and field effect transistors to introduce time dependent parameters into  $G(s)$ . Means for a practical solution of high-order, time-variable equations of the type which will be encountered in test stand work are being studied.

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3. Ibid., Figure 15, p. 59.
4. Ibid., Figure 13, p. 55.
5. "Basic Hydrostatic Pad Design," J. Boyd, H. N. Kaufman, A. A. Raimondi; Journal of Am. Soc. of Lubrication Engineers; Part I, September 1965, p. 391ff; Part II, October 1965, p. 439ff.
6. Ibid., Figure 2-C.
7. NASA Report No. N63-20023, "Investigation of Thrust Compensation Methods," J. A. Sprouse and W. X. McGregor.

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